CHAPTER 13

COMPRESSORS

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13.1 INTRODUCTION

A compressor is a machine for compressing gas from an initial intake pressure to a higher exhaust pressure through a reduction in volume. A compressor consists of a driving unit, the compression unit and accessory equipment. The driving unit provides power to operate the compressor and may be an electric motor or a gasoline or diesel engine. Types of gases compressed include air for compressed tool and instrument air systems; hydrogen, oxygen, etc. for chemical processing and various gases for storage or transmission. A compressed air system consists of one or more compressors, each with the necessary power source, air regulator, intake air filter, aftercooler, air receiver, and connecting piping, together with a distribution system to carry the air to points of use.

Compressors can be classified, in their broadest sense, in two categories: (1) positive displacement and (2) centrifugal. The positive-displacement classification can generally be described as a "volume reducing" type. In essence, an increase in gas pressure can be achieved by simultaneously reducing the volume enclosing the gas. In all positive displacement compressors, a measured volume of inlet gas is confined in a given space and then compressed by reducing this confined volume. Next, the gas at this now elevated pressure is discharged into the system piping.

The centrifugal classification refers to the type of velocity increase for centrifugal action. In a centrifugal compressor the gas is forced through the impeller by rapidly rotating impeller blades. The kinetic velocity energy from the rotating impeller is converted to pressure energy, partially in the impeller and partially in the stationary diffuser. The stationary diffuser converts the velocity head into pressure.

Each type of compressor is designed for specific applications and requirements. A reliability analysis therefore requires an investigation of the design features for the particular compressor. It is important to know what is inside the compressor not only to know the failure rate, but also how to logistically support the compressor in terms of spare parts and maintenance philosophy. The following section provides a basic description of the different types of compressors.

13.2 POSITIVE DISPLACEMENT COMPRESSORS

Positive displacement compressors include a wide spectrum of design configurations. As shown in Figure 13.1, positive displacement machines can be further defined by two sub classifications: rotary and reciprocating.

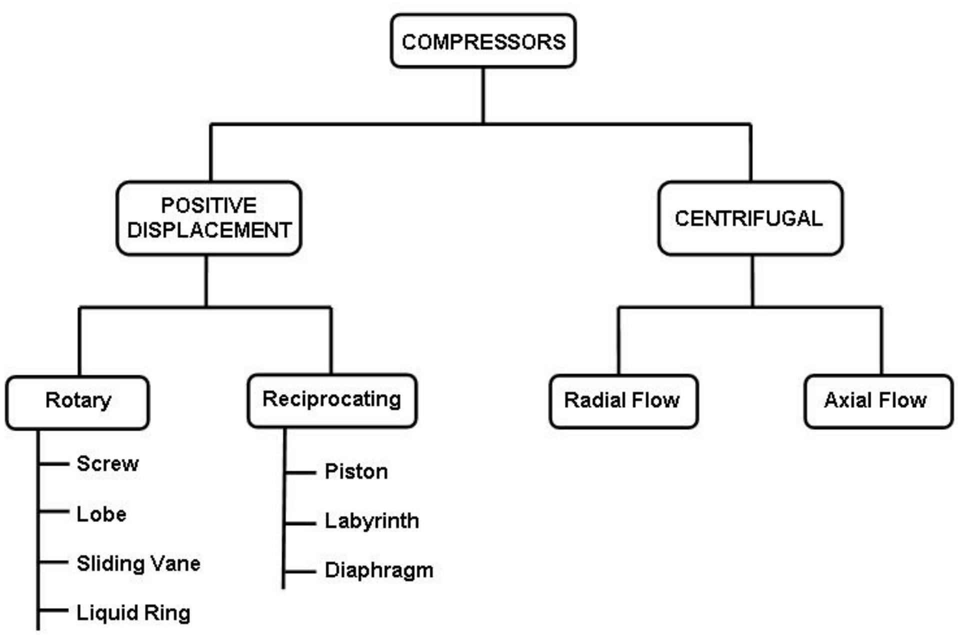


Figure 13.1 Common Classifications for Compressors

13.2.1 Rotary Compressors

Rotary positive displacement compressors incorporate a rotating element to displace a fixed volume of gas during each machine revolution. The following paragraphs provide a brief description of the different types of rotary compressors.

Rotary Screw - A common rotary compressor is the rotary screw. Rotary screw compressors produce compressed gas by filling the void between two helical mated screws and their housing. As the two helical screws are turned, the volume of gas is reduced resulting in an increase of gas pressure. Cooling and lubrication are obtained by injecting oil into the bearing and compression area. After the compression cycle, the oil and gas are separated before the gas is exhausted from the compressor.

Lobe - The rotary lobe compressor is typically constructed with two or three figure eight-shaped rotors, meshed together, and driven through timing gears attached to each shaft. It is a relatively low pressure machine (normally 5 to 7 psig and up to 25 psig for special types) and is well suited for applications with vacuum pressures. A lobe compressor provides a large throughput capability with little or no flow pulsation.

Sliding Vane - The sliding vane rotary compressor has a rotor construction which is offset, containing slots for vanes to slide in and out during each revolution. As the rotor turns during a single revolution, compression is achieved as the volume goes from a maximum at the intake ports to a minimum at the exhaust port. The vanes are forced outward from within the rotor slots and held against the stator wall by rotational acceleration. Oil is injected into the gas intake and along the stator walls to cool the gas, lubricate the bearings and vanes, and provide a seal between the vanes and the stator wall. After the compression cycle, the oil and gas are separated prior to the gas being transferred from the compressor.

Liquid Ring – In a liquid ring compressor the rotor is positioned centrally in an oval-shaped casing. During rotation, which happens without metal-to-metal contact, a ring of liquid is formed which moves with the rotor and follows the shape of the casing. During rotation, the liquid completely fills the chambers of the rotor and as the rotation continues, the liquid follows the contour of the casing and recedes again, leaving spaces to be filled by the incoming gas. As a result of the suction action thus created, gas is pulled into the compressor. As the rotation progresses, the liquid is forced back into the chambers, compressing the gas. This gas is forced out of the discharge port through an outlet flange. The compressor is fed continuously with liquid which maintains a seal between the inlet and discharge ports and at the same time removes the heat of compression. This liquid leaves the compressor together with the compressed gas and is separated from the gas in a discharge separator.

13.2.2 Reciprocating Compressors

Reciprocating air compressors are positive displacement machines in that they increase the pressure of the air by reducing its volume. As shown in Figure 13.1, there are various reciprocating compressor designs, the most common being the piston.

Piston - In this design, successive volumes of air are taken into the compressor and a piston within a cylinder compresses the air to a higher pressure. Air is released by mechanical valves that typically operate automatically by differential pressures. Inlet valves open when the pressure in the cylinder is slightly below the intake pressure. Discharge valves open when the pressure in the cylinder is slightly above the discharge pressure. Depending on the system design, cylinders may have one or multiple suction and discharge valves. Single stage compressors are commonly available for pressures in the range of 70 psi to 100 psi and two stage compressors are generally used for higher pressures in the range of 100 psi to 250 psi. A reciprocating air compressor is single acting when the compression is accomplished using one side of the piston and double acting if compression is accomplished using both sides of the piston during the advancing and retreating stroke.

Compression to high pressures in a reciprocating compressor may result in a temperature rise too great to permit the compression to be carried to completion in one cylinder, even though it is cooled. In such cases, the compression is carried out in stages, with a partial increase of pressure in each stage, and cooling of the gas between stages. Two and three-stage compression is common where pressures of 300-1000 psi are needed. In determining the number of stages (pistons) within a reciprocating compressor, the change in temperature across a stage, loading of the piston rod, and change in pressure across a stage are among the parameters taken into consideration.

Labyrinth - The labyrinth compressor is a vertical type reciprocating machine. In this type of compressor, rider rings and piston rings are not used as in the case of a horizontal type design. In labyrinth piston compressors, an extremely large number of throttling points provide the sealing effect around pistons and piston rods. No contact seals are used. The piston contains a labyrinth type piece at the center called a skirt. The cylinder also contains serration-like labyrinths on its inside surface. The piston is not in direct contact with the cylinder and close clearance is maintained between the piston and cylinder.

Labyrinth compressors are used where total dry operation is required and where lubricants are not allowed in the cylinders such as an oxygen compressor where safety is extremely important. Labyrinth compressors are also employed in applications where the process gas is heavily contaminated with impurities.

Diaphragm - The diaphragm compressor is a unique design employing a flexible diaphragm to compress the gas. The back and forth moving membrane is driven by a rod and a crankshaft mechanism. Only the membrane and the compressor box come in contact with the pumped gas and thus the diaphragm compressor is often used for pumping explosive and toxic gases. The membrane has to be reliable enough to take the strain of pumped gas. It must also have adequate chemical properties and sufficient temperature resistance.

Piston and diaphragm compressors possess many of the same components: crankcase, crankshaft, piston, and connecting rods. The primary difference between the two compressor designs lies in how the gas is compressed. In a piston compressor, the piston is the primary gas displacing element. However, in diaphragm compressors compression is achieved by the flexing of a thin metal, rubber or fabricated disk which is caused by the hydraulic system and operated by the motion of a reciprocating piston in a cylinder under the diaphragm. The diaphragm completely isolates the gas from the piston during the compression cycle. A hydraulic fluid transmits the motion of the piston to the diaphragm.

Diaphragms, in general, are round flexible plates which undergo an elastic deflection when subjected to an axial loading. In the application of compressors, this axial loading and elastic deflection creates a reduction in volume of the space adjacent to the diaphragm. The gas is compressed and a pressure builds. The diaphragm can be designed in many different ways with variations in such parameters as materials, size and shape.

13.3 CENTRIFUGAL COMPRESSORS

Centrifugal compressors depend on the transfer of energy from a rotating impeller to a gas discharge. The centrifugal force utilized by a centrifugal compressor is similar to a centrifugal pump. As gas enters the eye of the impeller the rotating impeller presses the gas against the compressor casing. The high speed spinning impellers accelerate the gas as additional gas is pressed against the casing by the impeller blades. A liquid ring (or piston) rotary is constructed of circular vanes, turning inside a casing sealed with a liquid. Centrifugal forces cause the liquid to form a ring around the periphery of the casing interior, while forcing the gas inward toward the center of the vaned rotor. The gradual decrease in volume increases the pressure of the gas. Any liquid entrained in the gas is separated out. This type of compressor is characteristically used in low pressure and vacuum applications. Centrifugal compressors are normally designed for higher capacity than positive displacement machines because flow through the compressor is continuous. Typical applications include aircraft engines.

Centrifugal compressors can be divided into two subcategories based on the direction of flow of the product gas: radial flow and axial flow machines. The characteristic curves of these machines offer a wide range in flow with a corresponding small change in head pressure. The centrifugal compressor is a continuous duty compressor with few moving parts making it well suited to high volume applications. The lack of rubbing parts in the compressed gas stream is a particularly desirable feature of these machines from a reliability standpoint.

Radial Flow - In radial compressors, velocity is imparted to a gas stream through centrifugal forces acting in a radial direction to the shaft. The simplest style of radial centrifugal compressor is the single-stage overhung design. The conventional closed or shrouded impeller is used for adiabatic heads to about 12,000 ft-lb/lb. The open, radial-bladed impeller develops more head for the same diameter and speed.

Axial Flow - In axial flow machines, the gas flow remains parallel to the shaft, without a direction change. These machines are typically used for higher capacities than radial flow machines, but generate much lower head pressure per stage. As a result, these machines are usually built with many stages. The characteristic performance curve is steeper than that of radial flow machines, with a more narrow stability range.

In summary, the different types and designs of compressors will result in different failure modes and failure rates. The next section provides some failure modes, causes and effects that need to be considered prior to estimating the total failure rate of the compressor in its operating environment.

13.4 COMPRESSOR FAILURE MODES

Figure 13.1 shows the various types of available compressor designs. Within these nomenclatures there are specific compressor designs with there own failure modes. To obtain an accurate list of failure modes for an individual compressor, a detailed parts list is needed and a thorough analysis of the interaction of component parts is required. For example, several stages of compressor units may be included in the overall compressor system which will require the determination of the effects of failure of adjacent stages if there is a failure of one particular stage.

Failure modes for compressors and certain compressor parts are listed in Table 13-1. Some failure modes are more prevalent than others as a direct result of the variety of compressor types and differing environmental conditions of operation.

Table 13-1. Compressor Failure Modes
(References 2 and 86)

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
General compressor failure modes (see specific compressor type failure modes below)		
Seal failure	See Chapter 3	- Reduced output
Bearing failure	See Chapter 7	- Low flow pulsation
Gear failure	See Chapter 8	- Compressor noise, vibration and complete failure
Belt failure	See Chapter 21	
Shaft failure	See Chapter 20	
Clogged filter assembly	- Contaminants - See Chapter 11 for additional failure causes	- Corrosion, excessive temperature causing winding damage
Temperature sensor failure	See Chapter 19	- Loss of overload protection
Loss of motor power source	- Compressor overload - Misalignment between motor and compressor - See Chapter 14 for additional failure causes	- Loss of compressor output - Contaminants from burnt motor windings
Corrosion, water hammer, freeze damage	- Moisture within the compressor - Discharge temperature <55 C higher than inlet air	- Acid gases mixed with internal moisture accelerates the corrosion process
Internal corrosion	- Faulty filtration - Acid gases from the environment	
Clogged suction strainer	- Mechanical wear - Erosion of close-clearance moving parts	- Loss of gas capacity
Pressure pulsations		
Loss of gas output	- Fractured compressor casing - Seal leakage - Connection failure - See individual compressor types below	- Compressor failure
Improper lubrication of mechanical parts	- Seal leakage	- Reduced compressor efficiency - Eventual loss of output

Table 13-1. Compressor Failure Modes (Continued)

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Increased friction and wear	- Contaminants - Poor lubrication - Excessive heat from running compressor outside limits	- Decreased performance, - Increased vibration
Enclosure damage caused by vibration	- High fluctuating stresses - Insufficient foundation	- Material fatigue
The following failure modes apply to reciprocating compressors and need to be considered in addition to the general compressor failure modes above		
Cylinder fails to move	- Spring loaded valve fails to open	- Loss of gas output
Cylinder leakage	- Mechanical wear - Damaged seal	- Reduced compressor efficiency
Liquid entering one or more compression cylinders	- Seal failure	- Permanent valve damage - Compressor failure
Damaged piston rings	- Low compressor oil - Wear	- Maintenance required
Inoperative suction on discharge valve	- Valve leakage - Discharge valve fails to open	- Reduced compressor efficiency
Damaged cylinder packing ring	- Moisture entering cylinder	- Permanent valve damage - Reduced compressor efficiency
Damaged piston – crankshaft connecting rod	- Mechanical binding - Loss of lubricant	- Noisy compressor - Reduced compressor efficiency
Unbalanced crankshaft	- See Chapter 20	- Noisy compressor - Reduced compressor efficiency
Mechanical overloading of piston	- Low compressor oil - Wear - Excessive duty cycle	- Compressor failure

Table 13-1. Compressor Failure Modes (Continued)

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
The following failure modes apply to rotary compressors and need to be considered in addition to the general compressor failure modes above		
Variable clearance between helical rotors	- Bent shaft, shock, vibration	- Severe rotor wear - Compressor failure
Contaminants between helical rotors	- Compressor operated in contaminated environment - Acid in ambient air	- Severe rotor wear - Compressor failure
Loss of lube film between helical rotors	- Loss of lubricant - Worn rotors	- Loss of gas capacity
Loss of coolant capability	- Loss of injected lubricant	- Damaged rotors - Compressor failure
Clogged discharge port	- Clogged filter - Contaminants through rotor section	- Loss of compressor air output
Oil and gas not sufficiently separated in high pressure area	- Filter damage - Plugged filter	- Contaminated gas output
Leakage of output gas back to low pressure side (slippage)	- Mechanical wear	- Loss of gas capacity
Reduction of internal clearances	- Distortion of rotor due to cyclic loading;	- Rubbing, increased wear
Accumulation of water in the lubricant	- Cooler failure	- Early compressor failure
Valve sticking	- Over lubrication, moisture in oil	- Overheating, increased wear
The following failure modes apply to diaphragm compressors and need to be considered in addition to the general compressor failure modes above		
Accelerated curing, embrittlement of diaphragm	- Extreme high or low temperature	- Decreased performance
Corrosion or cracking of diaphragm	- Contaminants	- Decreased performance
Valve sticking	- Over lubrication, moisture in oil	- Overheating, increased wear

Table 13-1. Compressor Failure Modes (Continued)

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Diaphragm rupture or crack	- Inadequate strength characteristics - Insufficient material pasticity - Corrosion - Local stresses at seating surface - Material hardening	- Loss of air output

13.5 FAILURE RATE MODEL FOR COMPRESSOR ASSEMBLY

A compressor system is made up of one or more stages. For example, a single-stage compressor is comprised of a single element or group of elements in parallel. However, two and three-stage compression may be required where higher pressures are needed. The total compressor may therefore be comprised of elements or groups of elements in series to form a multistage compressor based on the change in temperature and pressure across each stage.

Every compressor to be analyzed for reliability will be a unique design and comprised of many different components. The following example equation will need to be modified for the particular compressor design.

$$\lambda_C = (\lambda_{FD} \bullet C_{SF}) + \lambda_{CA} + \lambda_{BE} + \lambda_{VA} + \lambda_{SE} + \lambda_{SH} \qquad \textbf{(13-1)}$$

Where: λ_C = Total failure rate of compressor, failures/million hours

λ_{FD} = Failure rate of fluid driver, failures/million hours (See Section 13.7)

C_{SF} = Compressor Service Multiplying Factor (See Table 13-5)

λ_{CA} = Failure rate of the compressor casing, failures/million hours (See Section 13.6)

λ_{BE} = Total failure rate of compressor shaft bearings, failures/million hours (See Chapter 7)

λ_{VA} = Total failure rate of control valve assemblies, failures/million hours (See Chapter 6)

λ_{SE} = Total failure rate of compressor seals, failures/million hours (See Chapter 3)

λ_{SE} = Failure rate of compressor shaft, failures/million hours (See Chapter 20)

Additional parameters in the equation that may be necessary include timing gears (Chapter 8), belt (Chapter 21), couplings (Chapter 17), sensors (Chapter 19), filter (Chapter 11) and clutch (Chapter 12).

Different compressor configurations such as piston, rotary screw and centrifugal have different parts within the total compressor and it is important to obtain a parts list for the compressor prior to estimating its reliability. For example, a reciprocating piston compressor will contain suction and discharge valves while a rotary screw compressor will not normally contain control valves but may contain some unique oil/gas separator filters. The fluid driver of the compressor may be only a small part of the total compressor failure rate. The failure rate for each part comprising the compressor must be determined before the entire compressor assembly failure rate, λ_C, can be determined. Section 13.7 provides some guidance as to the various compressor configurations. Failure rates for each part will depend on expected operational and environmental factors that exist during normal compressor operation. It is important to consider each compressor stage in a multi-stage compressor as a separate compressor with the total failure rate as the sum of the failure rates for the individual stages. For example, a total compressor system may be comprised of a centrifugal compressor followed by an oil-free reciprocating compressor. The total system failure rate is the sum of the failure rates for the two units.

13.6 FAILURE RATE MODEL FOR CASING

The compressor casing, normally a very reliable component, can have a large effect on the life of other components in the compressor assembly (especially seals and bearings). Casing construction design, connections, casing material and production testing are specified to meet the application, safety and casing-pressure requirements. The value of reliability of compressor casings, through the experience of many different manufacturers, can generally be equated to a λ_{CA} value of 0.010 failures/million hours.

13.7 FAILURE RATE MODEL FOR DESIGN CONFIGURATION

Various reliabilities are inherent in specific designs and types of compressors. For example, it is expected that the reliability due to wear will be different in a rotary screw compressor compared to a centrifugal compressor due to the nature of metal-to-metal contact and rotor speeds. The various chapters of this handbook can be used to estimate the failure rates of the individual component parts. The fluid driver parameter λ_{FD} can be approximated by data presented in Table 13-4 for various types of fluid drivers, developed from information collected by such sources as OREDA and the U.S. Navy.

Typical component parts of a compressor that need to be evaluated for reliability include the following:

- drive unit
- air end
- cooling system
- receiver tanks
- air dryers
- filters
- piping distribution system

A rotary compressor is typically comprised of the following components:

- air intake filter unit usually a two stage assembly required by rotary compressors
- rotor unit comprised of screws, lobes or vanes to compress the gas and a discharge port. A non-lubricated rotary compressor will contain timing gears to drive the rotors
- oil injection unit to supply lubrication between the rotating screws or vanes and prevent gas leakage
- aftercooler unit
- oil/gas separator unit to separate the oil from the gas
- oil filter unit
- oil cooler unit

A reciprocating compressor is typically comprised of the following components:

- compressing unit consisting of air cylinders, single or double acting piston for each cylinder, air inlet valves and discharge valves
- mechanical unit consisting of connecting rods, piston rods, crossheads, and a crankshaft and flywheel
- lubricating unit for bearings, gears, and cylinder walls consisting of a pump or force-fed lubricator to supply oil to the compressor cylinders
- regulation unit to maintain the pressure in the discharge line and gas storage tank at the required pressure

A centrifugal compressor is typically comprised of the following components:

- Rotating components mounted on a shaft that drives a central drum retained by bearings

13.7.1 Compressor Service Load Multiplying Factors

As mentioned previously in this Chapter, each type of compressor is uniquely designed for a particular application and requirement. Some designs are more sensitive to a particular operating environment and require particular attention as part of the reliability estimate. For example, inlet air condition is an important consideration in estimating compressor reliability. A proper inlet filtration prevents erosion of seals and close clearance moving parts. Some compressor designs are more sensitive than others to the entrance of contaminants into the compressor. Another factor is high ambient humidity that if not controlled can cause corrosion. Duty cycle, ambient operating temperature, lubrication quality, shock and vibration are all operating and environmental parameters that need to be considered with respect to the sensitivity of the design to that parameter. Table 13-5 provides multiplying factors that modify the base failure rate considering each potential operating or environmental condition.

13.8 DIAPHRAGM FAILURE RATE MODEL

The diaphragm compressor is a unique design employing a flexible diaphragm to establish compression of gas. The back and forth moving membrane is driven by a rod and a crankshaft mechanism as shown in Figure 13.3. Only the membrane and the compressor box come in contact with the pumped gas and thus the diaphragm compressor is often used for pumping explosive and toxic gases. Because of the unique design properties and applications of the diaphragm compressor this section is included to provide a more detailed analysis of the diaphragm itself. Table 13-4 can be used in lieu of this detailed procedure.

λ_{FD} in Equation (13-1) for the diaphragm compressor can be written as:

$$\lambda_{FD} = \lambda_{DI}$$

Where:

λ_{DI} = Total failure rate of the diaphragm assembly

And:

$$\lambda_{DI} = \lambda_{BFD} \bullet C_P \bullet C_{AC} \bullet C_{LC} \bullet C_T \qquad \textbf{(13-2)}$$

Where:

λ_{BFD} = Compressor diaphragm base failure rate, 0.58 failures/million hrs.

C_P = Factor for effects of axial loading (See Section 13.8.1)

C_{AC} = Factor for effects of atmospheric contaminants (See Section 13.8.2)

C_{LC} = Factor for effects of liquid contaminants (See Section 13.8.3)

C_T = Factor for effects of temperature (See Section 13.8.4)

13.8.1 Axial Load Multiplying Factor

Diaphragms, in general, are round flexible plates which undergo an elastic deflection when subjected to an axial loading. In the application of compressors, this axial loading and elastic deflection creates a reduction in volume of the space adjacent to the diaphragm. The gas is compressed and a pressure builds. The diaphragm can be designed in many different ways with variations in such parameters as materials, size and shape. The model developed for a compressor diaphragm is shown in Figure 13.2. It has a passive area in the center which is rigid. This area transmits a force from the push rod to the diaphragm. To be effective, the thickness of the rigid center should be at least 6 times the thickness of the diaphragm.

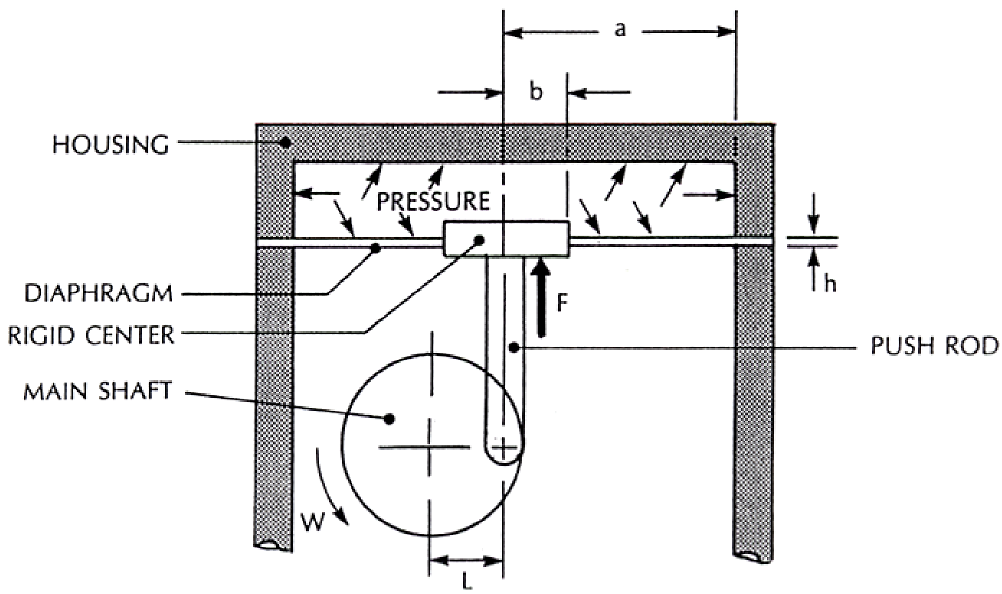


Figure 13.2 Compressor Diaphragm Model

The characteristic equations describing the compressor diaphragm are given in Equations (13-3) through (13-9) and are based on the following restrictive assumptions:

(1) Diaphragm is flat and of uniform thickness.

(2) Diaphragm material is isotropic and homogeneous.

(3) All forces, loads, and reactions are applied normally to the plane of the plate.

(4) Diaphragm thickness not greater than 20% of its diameter.

(5) The effects of shearing stresses and pressures on planes parallel to the surface of the diaphragm have not been taken into account. They are considered insignificant in diaphragms with thickness to radius ratios (*h/a*) of less than 0.15.

(6) The stresses created in a diaphragm due to bending and tensile loading may be combined by summing their values (method of superposition).

The characteristic equation of a rigid center diaphragm loaded by a force for any magnitude of deflection is given by Equation (13-3). It is applicable for (b/a) ratios greater than 0.05.

$$F = \frac{\pi E}{a^2}\left[\frac{h^3 y_o}{K_F} + h\, y_o^{\,3} B\right] \qquad \textbf{(13-3)}$$

Where: F = Force applied to rigid disk of diaphragm, lb

E = Modulus of elasticity, lbs/in^2

a = Radius of diaphragm, in

h = Diaphragm thickness, in

y_o = Vertical deflection at center of diaphragm, in

K_F = Modified Stiffness Coefficient based on diaphragm

bending loads, $$= \frac{3\left(1-\eta^2\right)}{\pi}\left[\frac{c^2-1}{4c^2} - \frac{\ln c^2}{c^2-1}\right] \qquad \textbf{(13-4)}$$

B = Stiffness coefficient based on diaphragm tensile loading, as follows:

$$= \frac{\dfrac{7-\eta}{3}\left(\dfrac{1+b^2}{a^2} + \dfrac{b^4}{a^4}\right) + \dfrac{\left(3-\eta\right)^2 b^2}{\left(1+\eta\right) a^2}}{\left(1-\eta\right)\left(1-\dfrac{b^4}{a^4}\right)\left(1-\dfrac{b^2}{a^2}\right)^2} \qquad \textbf{(13-5)}$$

η = Poisson's ratio

c = Ratio of radii (diaphragm-to-disk), a/b, in/in

b = Radius of rigid center plate of diaphragm, in

The maximum radial stress for a force-loaded diaphragm with rigid center occurs at the inner perimeter of the diaphragm (b):

$$\sigma_r = \frac{F\,K_F\,B_F}{2\,\pi\,h^2} \tag{13-6}$$

Where: σ_r = Maximum radial stress, lbs/in^2

B_F = Modified stiffness coefficient, based on diaphragm tensile loading

$$= \frac{2}{1-\eta^2}\;\frac{c^2\left(2c^2\ln c - c^2 + 1\right)}{\left(c^2-1\right)^2 - 4c^2\ln^2 c} \tag{13-7}$$

At equilibrium, where the force transmitted by the push rod in Figure 13.2 generates a maximum pressure in the chamber above the diaphragm (i.e., the rod has completed its stroke), a balance of forces in the vertical direction is established.

If the increased performance of a compressor is to be evaluated and the change in shaft power requirements are known, the following equation, in combination with Equation (13-6), can be used to evaluate the maximum induced stress in the diaphragm:

$$\sigma_r = \frac{396{,}000\,hp\,K_F\,B_F}{2\,\pi\,L\,\omega\,h^2} \tag{13-8}$$

Where: hp = Shaft output horsepower

L = Offset of eccentric shaft, in

ω = Output shaft speed, rpm

The maximum stress is calculated from Equation (13-8) for the compressor rated condition. Then the maximum stress for the actual operating condition is calculated in the same manner.

Empirical studies show that for moderate to high strains, a mechanical tearing of rubber, referred to as "mechanical-oxidative cut growth", can be the mechanism of failure for rubber diaphragms. The cut growth may greatly increase in the presence of oxygen. For this mode of failure, the fatigue life is inversely proportional to a power of the strain energy of the rubber. The strain energy is a characteristic of each type of rubber, and in turn, inversely proportional to the strain experienced by rubber under cyclic stressing. Figure 13.3 shows the stress-strain relationship for natural rubber compounds. Unlike many other engineering materials, rubber can be manufactured with a wide range of elastic moduli. Stiffness variations can be attained with no dimensional changes by varying the incorporation of fillers (reinforcing carbon blacks). This "hardness" variable is essentially a measurement of reversible elastic penetration (International Rubber Hardness Degrees or IRHD).

The stress developed in a rubber diaphragm can be calculated from Equation (13-6). Although rubber is flexible, (i.e., has low elastic and shear moduli), it is highly incompressible in bulk and its Poisson's ratio, η can be approximated as 0.5. This will facilitate the use of these equations. From the stress calculated, Figures 13.3 and 13.6 provide a corresponding load multiplying factor, C_P.

The value for strain obtained from Figure 13.3 must exceed 75%. Below this strain, the mechanical-oxidative cut growth mode of failure does not apply, and the C_P factor becomes 1.0.

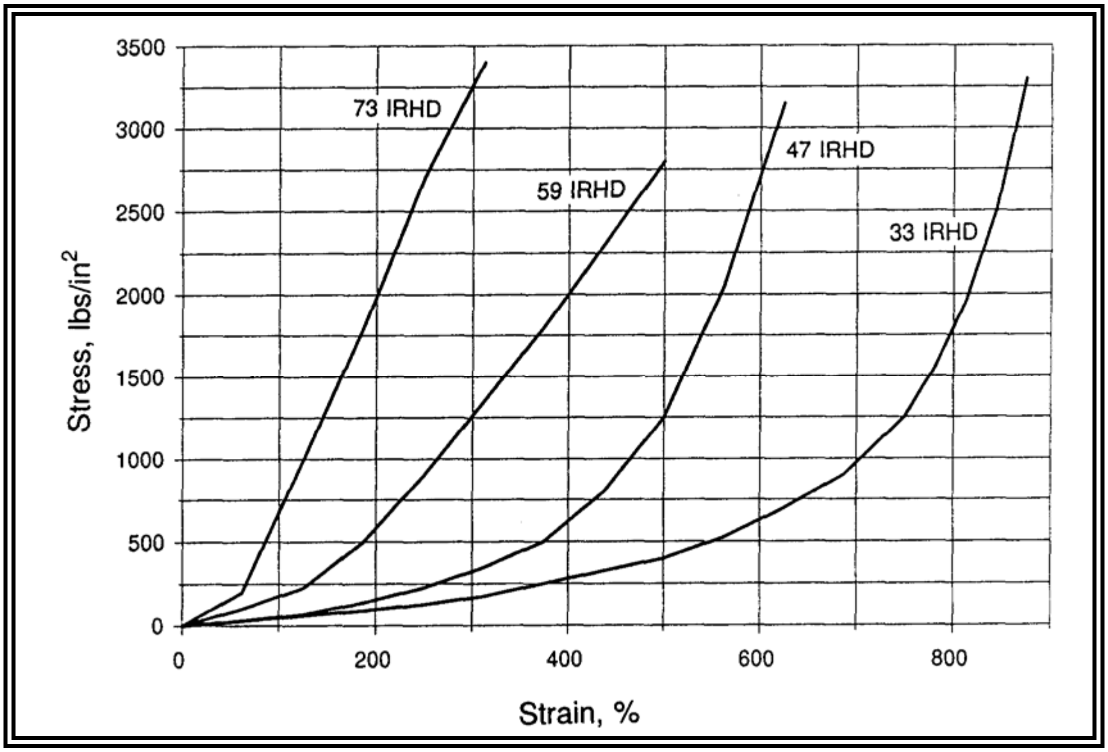


Figure 13.3 Tensile Stress-Strain Curves for Four Natural Rubber Compounds of Different Hardness (Ref 31)

13.8.2 Atmospheric Contaminant Multiplying Factor

The very small concentration of ozone in the atmosphere, normally a few parts per hundred million at ground level, may cause cracking in strained rubber components. Under cyclic conditions of strain below about 75%, ozone cut growth is the major factor in determining fatigue life.

Experimental data presented in Figure 13.7 illustrates that fatigue life is proportional to the concentration of ozone. The stress developed in a rubber diaphragm can be calculated from Equation (13-6). Poisson's ratio, η, can be equated to 0.5.

Table 13-2 can be used to determine the strain by dividing the value of stress obtained from Equation (13-6) by Young's modulus. Figure 13.7 and this strain value are then used to determine the contaminant air performance multiplying factor, C_{AC}.

13.8.2.1 Adjustment to Atmospheric Contaminant Multiplying Factor

In ozone-dominant failure potentials, the use of chemical anti-ozonant (coating) on the surface of the rubber diaphragm can reduce crack growth by a factor of 3. If a coating is used, the multiplying factor, C_{AC}, obtained from Figure 13.7 should be multiplied by 1/3.

13.8.3 Liquid Contaminant Multiplying Factor

Water absorption does not usually cause any significant deterioration of rubber, but the absorption of oil and solvents cause rubber to swell with a consequent deterioration in certain properties. Thin components can be expected to fail rapidly if the major surfaces are exposed to oil. Thick components are effectively protected by their bulk. Such components can last many years in an oily environment. Diffusion theory predicts that the mass of liquid absorbed per unit area of rubber (in the early stages of swelling) is proportional to the square root of the time taken for the absorption.

The rate of movement of the boundary between swollen and unswollen rubber is calculated from:

$$PR = \frac{L}{\sqrt{t}} \tag{13-9}$$

Where: PR = Penetration rate, in/sec$^{0.5}$

L = Depth of the swollen layer, in

t = Time that a given mass of liquid is absorbed by a given surface, sec

The failure rate for a rubber diaphragm is dependent on the presence of liquid contaminants and the viscosity of the liquid in contact with it. Typical penetration rates are shown in Figure 13.4. Figure 13.4 reveals that the penetration rate into natural rubber decreases as the viscosity of the swelling liquid increases.

An adjustment for various types of diaphragm materials can be made using the multiplying factors presented in Table 13-3. These factors should be multiplied by the penetration rate obtained from Figure 13.4 prior to using the nomograph in Figure 13.5.

Table 13-2. Hardness and Elastic Moduli

HARDNESS, IRHD	YOUNG'S MODULUS, E, lb/in^2
30	130
35	168
40	213
45	256
50	310
55	460
60	630
65	830
70	1040
75	1340

13.8.4 Temperature Multiplying Factor

The variations in ambient temperature commonly occurring in practice are unlikely to greatly affect fatigue behavior. Experiments over a range of -32 to 212 F indicate only a slight effect of temperature on the fatigue life of crystallizing natural rubber. In general, rubbers become weaker as the temperature is raised. There is a steady fall in strength up to a critical temperature at which an abrupt drop occurs. For natural rubber, this temperature is about 212 ^oF.

A temperature multiplying factor, C_T, can be developed as follows:

For: -32F < T $\leq$ 212 F, C_T = 1.0

and for: T > 212 F, C_T = 6.7

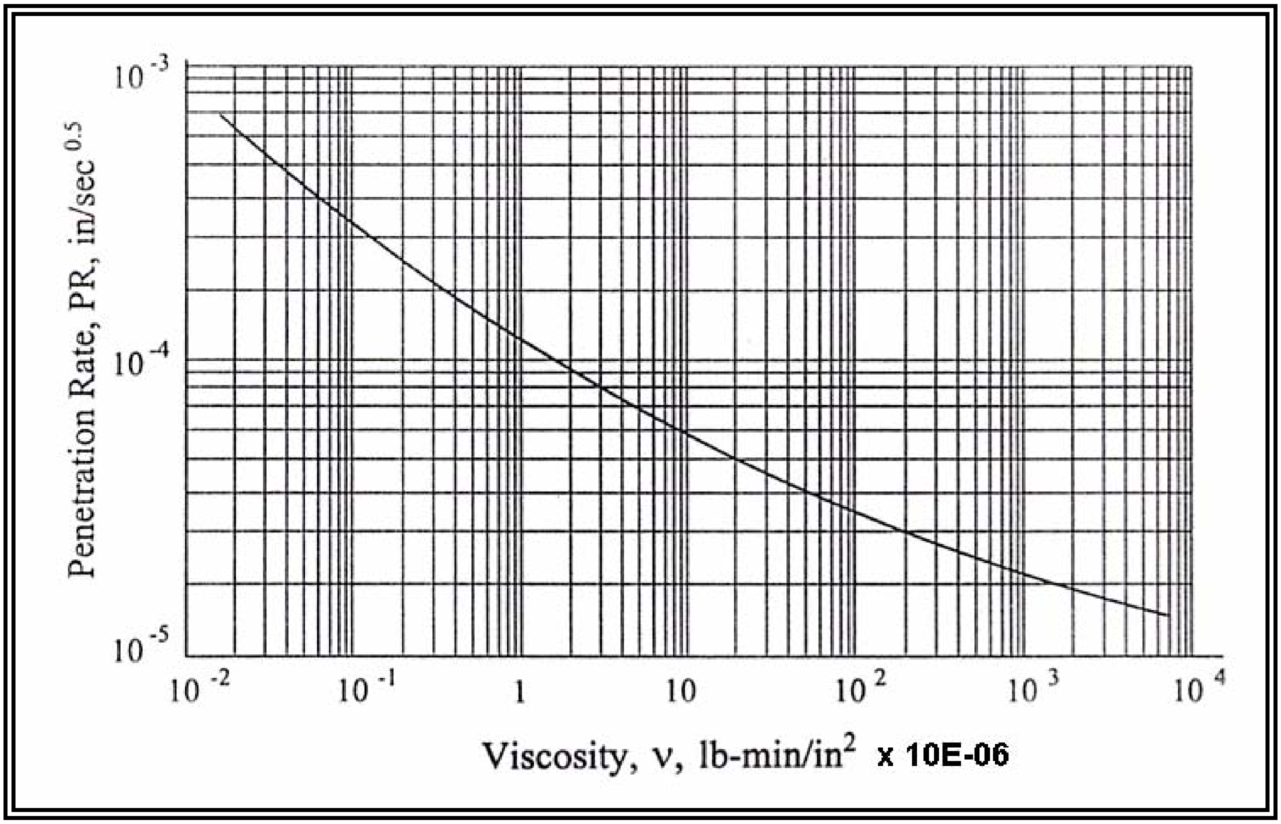


Figure 13.4 Effect of Liquid Viscosity on the Penetration Rate of Liquids into Natural Rubber

Table 13-3. Contaminant Adjustment Factor For Various Diaphragm Materials

RUBBER	X
Natural	1.0
Cis polybutadiene	1.3
Butyl	0.7
SBR	0.7
Neoprene WRT	0.4
Nitrile (38% acrylonitrile)	0.1
Metal	0.001

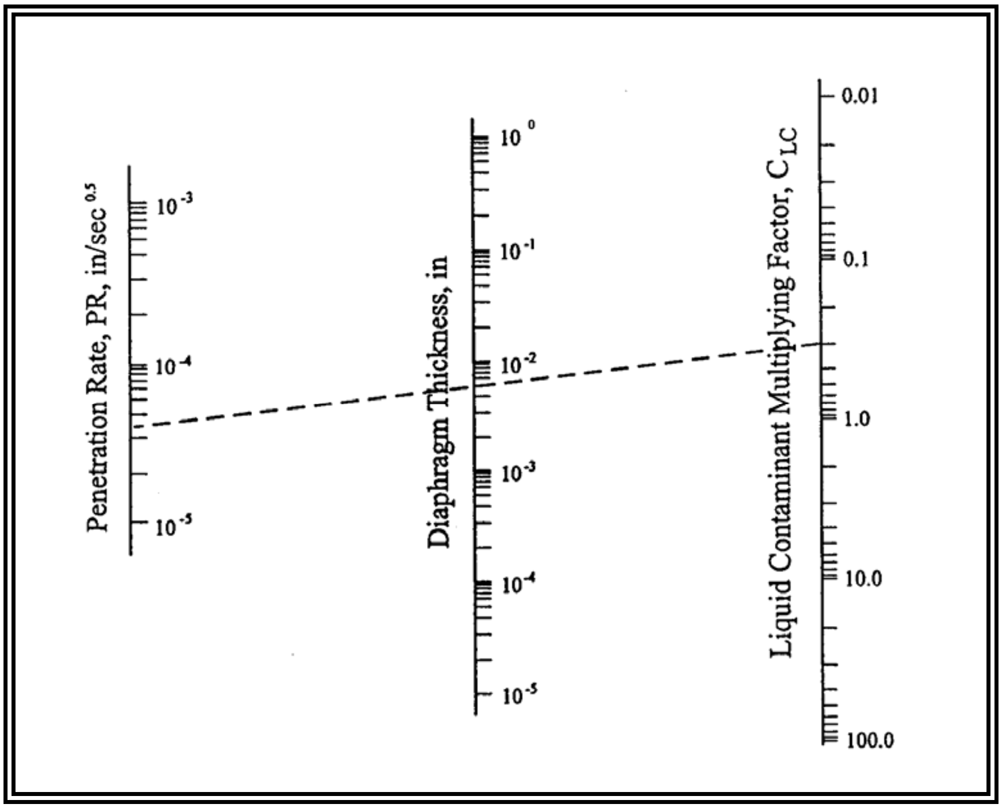
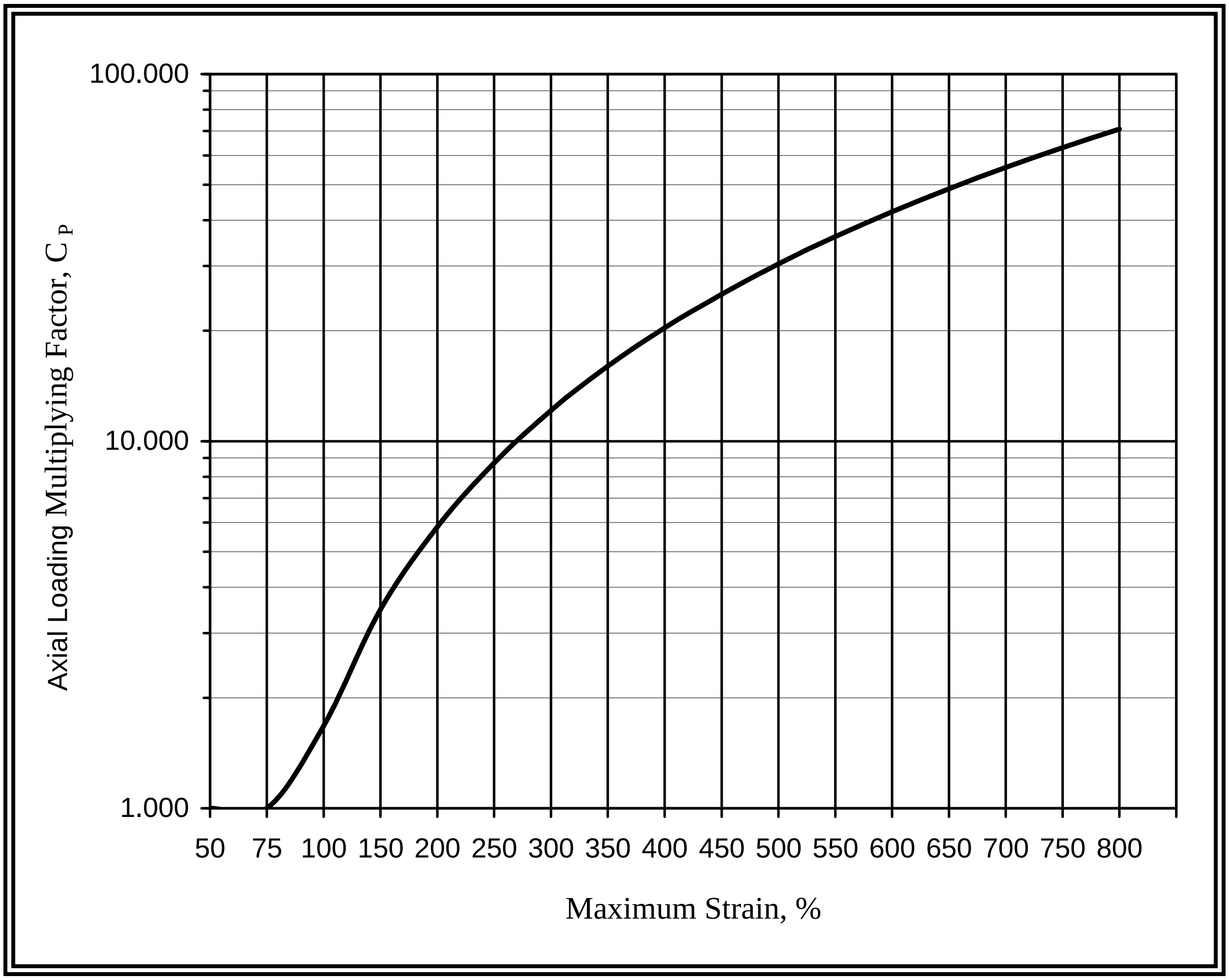


Figure 13.5 Nomograph for the Determination of Liquid Contaminant Multiplying Factor, C_{LC}

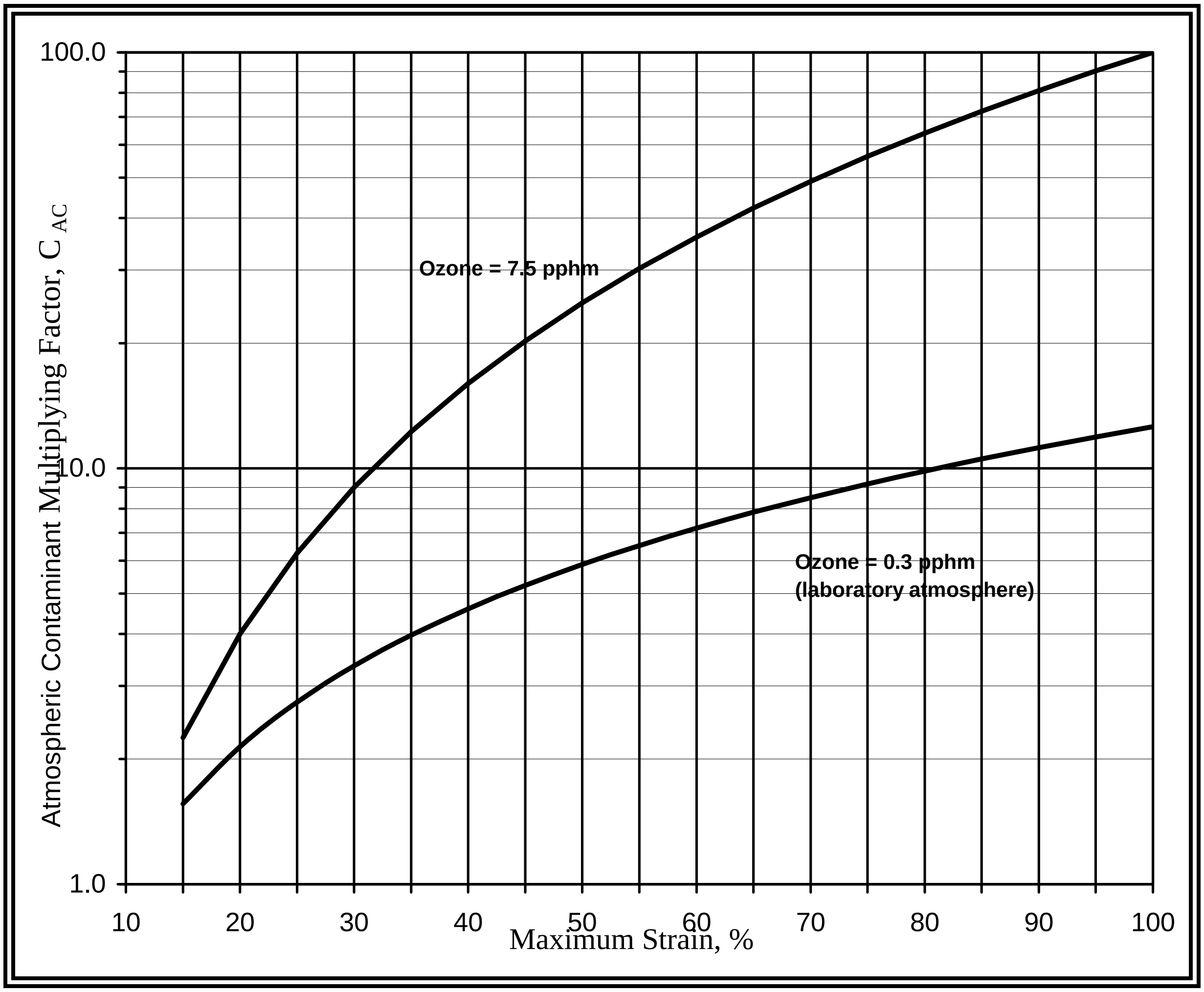


For $S \leq$ 75%: $C_P = 1.0$

For $S >$ 75%: $C_P = \left(\frac{S}{75}\right)^{1.8}$

Where: S = Strain, %

Figure 13.6 Axial Loading Multiplying Factor as a Function of Strain



For ozone 0.3 pphm: $C_{AC} = \left(\frac{S}{10}\right)^{1.1}$

For ozone 7.5 pphm: $C_{AC} = \left(\frac{S}{10}\right)^{2}$

Where S = Strain, %

Figure 13.7 Atmospheric Contaminant Multiplying Factor

Table 13-4. Failure Rate for Fluid Drivers (λ_{FD})

(*See note following table*)

FLUID DRIVER MODE	MODEL TYPE	BASE RATE* λ_{FD}
Radial flow	------	12.0
Axial flow	------	12.0
Reciprocating	Single piston	14.0
Reciprocating	Double acting piston	16.5
Reciprocating	Labyrinth	16.5
Reciprocating	Rubber Diaphragm **	22.8
Reciprocating	Metal Diaphragm	28.5
Rotary	Vane	12.0
Rotary	Screw	12.0
Rotary	Lobe	12.0
Rotary	Liquid Ring	12.0

* Failures/million hours of operation

** See Section 13.8 for specific failure rate calculations

Note: If the complete compressor has multiple stages determine the failure rate for each stage as an independent compressor and total the failure rates.

Table 13-5 Compressor Service Load Multiplying Factors

Multiplying Factor	Centrifugal	Rotary	Reciprocating	Diaphragm
Normal duty cycle, operating temperature and humidity, air cleanliness with proper filtration, lubrication quality, vibration and shock loading	1.0	1.0	1.0	1.0
High duty cycle (> 5 cycles per /hour)	1.2	1.2	1.4	1.2
Extreme operating temperatures	1.1	1.1	1.4	1.4
Non-scheduled lubrication check	1.1	1.2	1.1	1.2
High vibration level and/or heavy shock loading	1.2	1.4	1.3	1.5
Poor inlet air quality	1.1	1.4	1.1	1.3

13.9 REFERENCES

In addition to specific references cited throughout Chapter 13, other references included below are recommended in support of performing a reliability analysis of compressors.

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